ISSN 0040-6015, Thermal Engineering, 2012, Vol. 59, No. 2, pp. 106–112. © Pleiades Publishing, Inc., 2012.
Original Russian Text © A.E. Zaryankin, N.A. Zroichikov, A.N. Paramonov, V.V. Noskov, E.Yu. Grigor'ev, 2012, published in Teploenergetika.

Pressure Pulsations in the Turbine Steam-Admission Path and Their Influence on the Vibration State of the Turbine Control Valves

A. E. Zaryankin[a], N. A. Zroichikov[b], A. N. Paramonov[c], V. V. Noskov[c], and E. Yu. Grigor'ev[c]

[a] *Moscow Power Engineering Institute, ul. Krasnokazarmennaya 14, Moscow, 111250 Russia*
[b] *OAO Mosenergo's TETs-23 Cogeneration Station, ul. Montazhnaya 1/4, Moscow, 104497 Russia*
[c] *ZAO ENTEK, ul. Krasnokazarmennaya 17, str. 3, Moscow, 111116 Russia*

Abstract—Data on pressure pulsations in the valve box, downstream of the diffuser seat, and in the subsequent steam line are presented for the case of using external valves, and it is shown that the level of pulsations depends essentially on the geometrical characteristics of the steam admission path.

DOI: 10.1134/S0040601512020176

The turbine control valves, which are used as actuators of the turbine control system, are aimed to ensure reliable operation of the turbine at all possible loads. In addition, they serve as additional protection preventing destruction of the turbine in emergency situations.

Unfortunately, the regular valves of almost all turbine construction companies do not perform the above-mentioned functions in full scope because their components experience high dynamic loads caused by a high level of pressure pulsations in the valve flow path. As a result, self-oscillations appear in certain modes of operation, which are the most frequent factor causing destruction of valve stems.

It is commonly believed that the valve itself generates highly unsteady flow in its flow path. This follows, among other things, from direct measurements of pressure pulsations downstream of the diffuser seats of valves [1, 2], according to which pressure pulsations in the flow supplied to the control stage's nozzle vanes reach 10% of the live steam pressure. For turbines with the initial pressure equal to 23.7 MPa, this makes around 2.4 MPa [1]. With the control valves installed separately from the turbine casing, the dynamic loads arising with such pressure pulsations inevitably give rise to increased vibration of the entire steam admission system and the steam lines connected with it. In some cases, the level of these pulsations does not allow the turbine to be operated in its normal mode. In particular, when the K-1000-66 turbine produced by Scoda was put in operation at the Temelin power station, the entire power unit had to be operated at a load no more than 25% of its rated level due to high vibration of the turbine valves and steam lines [3]. High dynamic loads acting on the steam admission system are the main factor causing destruction of many elements used in control valves. Therefore, development of control valves introducing minimal disturbances in the steam flow and insensitive to pressure pulsations arising in unsteady flow is of great interest for practical applications.

This work is just devoted to development of such valves.

THE EXPERIMENTAL INSTALLATION AND THE SYSTEM FOR MEASURING PRESSURE PULSATIONS AND DYNAMIC FORCES ON VALVE STEMS

The main specific feature of an angle control valve is that its outlet section is placed at an angle of 90° with respect to the inlet of working fluid into the valve box (accordingly, the flow is turned through 90°). With such a turn, intricate unsteady flow with salient vortex structures arises even in smooth channels.

Therefore, the considerable pressure pulsations observed downstream of the valve arise both due to the design features of the valve and as a natural consequence of the flow being turned through 90° with respect to its initial direction.

For obtaining quantitative values of pressure pulsations resulting from turning the flow through 90°C, the authors of this paper carried studies on a simplest installation (Fig. 1), in which smooth angular cylindrical elbow *1* is attached on one side to an air receiver by means of flange *2* and on the other side, to small cylindrical chamber *4* via flange *3*. Replaceable diffusers *5* can be connected to the outlet part of chamber *4*. A version with a sharp-angle turn was considered along with a smooth turn of the flow in studying pressure pulsations in turning elbows. In chamber *4* we measured air pressure and temperature, and in the outlet section of diffusers we installed a low-inertia pressure gage, using which pressure oscillograms were

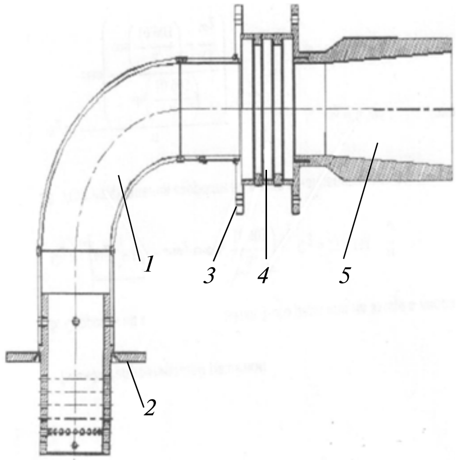

Fig. 1. Installation for studying pressure pulsations with flows turned through 90°.

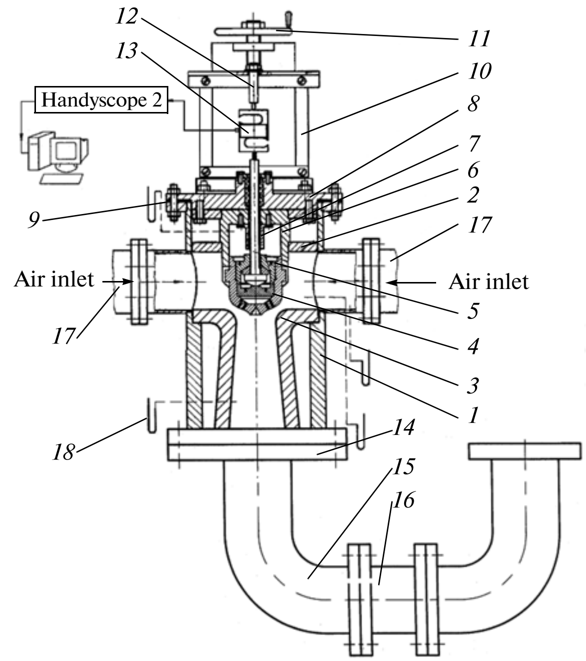

Fig. 2. Experimental setup for studying control valves.

recorded. Figure 2 shows the installation on which the control valves were studied. Cylindrical shell *1* houses partition *2* and diffuser seat *3*, which form a cylindrical valve box, in which studied valve *4* connected with stem *5* is placed. Casing *6* with sealing bush *7* simulating the protective casing of real valves is installed on partition *2*.

The upper part of cylindrical shell *1* is closed by cover *8* fitted with additional fluoroplastic seals *9*. Rod truss *10* with drive mechanism *11* connected to stem *5* of valve *4* is attached to the upper part of cover *8*. Strain-gage dynamometer *13* is placed in the gap between stem *5* and spindle of drive *12*. The dynamometer is connected to the Handyscop-2 measurement system the signal from which is transmitted to a personal computer. The lower part of shell *1* is connected through flanges *14* to pipeline *15*, the working fluid in which makes two turns through 90°. Insert *16* is installed in the pipeline after the first turn, which makes it possible to install special vortex suppressors downstream of the turn, which serve to reduce the pulsation level in the flow of working fluid admitted to the subsequent pipeline segment of to the control stage's nozzle vane. Compressed air is admitted to the installation via two pipelines *17*. One-sided supply of working fluid is organized by closing one of these pipelines by a solid diaphragm.

Pressures are measured by water manometers *18* and special DPS-011 pressure gages (produced by NIIFI in Penza), the outputs of which are connected to the Mera multifunctional measurement system. The flowrate of medium was determined using a calibrated Laval nozzle, and vibrations were measured by either an Agat portable vibration meter (produced by Diamekh, Russia) or by the movable measurement system produced by Pruftechnik (Germany).

The use of such installation makes it possible to obtain flowrate, force, and vibration characteristics for each valve.

RESULTS FROM THE STUDY OF A TURN ELBOW WITH SUBSEQUENT DIFFUSERS

The considered series of experiments is aimed at showing the level of disturbances arising in the system consisting of a turning bend and subsequent diffusers.

Figure 3 shows the change in the average relative amplitude of pressure pulsation $\Delta\bar{p}$ in the outlet section of diffusers with increasing their aperture angle ($\Delta\bar{p}$ is the ratio of the average amplitude of pressure pulsation in the flow to the full deceleration pressure upstream of the turning bend).

With the flow turned through a sharp angle (the diffuser aperture angle $\alpha = 0°$), the relative pulsation of pressure was equal to 8.6% of the working fluid's initial pressure. After installing a conical diffuser with the aperture angle $\alpha = 7°$, the amplitude of pressure pulsations downstream of it decreased to 7%. This points to the fact that the initial high level of pressure pulsations is reduced in separation-free diffusers. With the diffuser aperture angle increased to 10°, the relative amplitude of pressure pulsations increased to 8%, and with $\alpha = 30°$ its value reached 11%.

As should be expected, the amplitude of pressure pulsations became considerably smaller when a smooth turning of the flow was organized. In this case, the dependence $\Delta\bar{p} = f(\alpha)$, obtained in the outlet sec-

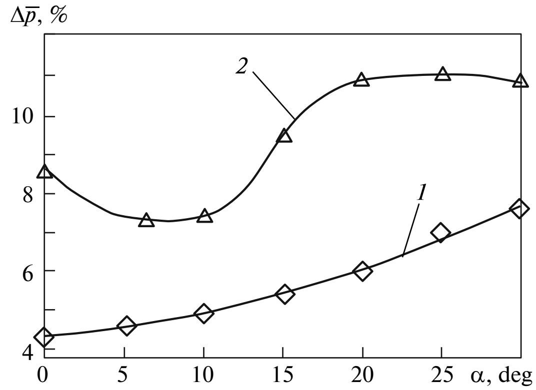

Fig. 3. Average relative amplitude of pressure pulsations $\Delta\bar{p}$ in the outlet section of a conical diffuser vs. its aperture angle α. (*1*) Smooth turning of the flow and (*2*) angular turning of the flow.

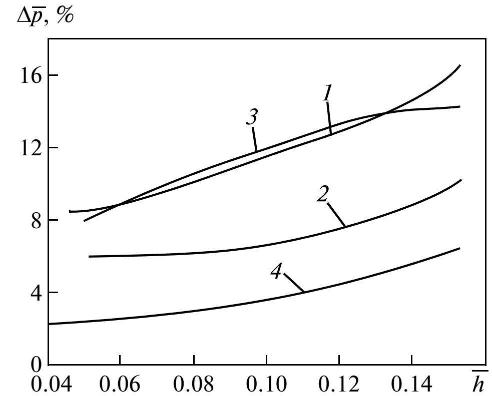

Fig. 4. Dependence of average relative amplitude of pressure pulsations $\Delta\bar{p}$ in the valve box (*1*) and (*2*) and downstream of the diffuser seat (*3*) and (*4*). Diffuser seat aperture angle α: (*1*) and (*3*) 10° and (*2*) and (*4*) 7°.

tion of the studied diffusers was monotonic in nature, and with α increased to 10°, the value of $\Delta\bar{p}$ increased by only 0.8%. The largest growth of $\Delta\bar{p}$ is observed with α > 10° when the flow in the diffuser channel begins to separate from the walls.

The following conclusions can be drawn from the presented data. First, it is shown that with the flow turned through 90° with respect to its initial direction, the relative pulsations of pressure reach the same level as that obtained downstream of the diffuser seats of full-scale control valves [1, 2]. Second, it was found that some decrease of pressure pulsations downstream of the diffuser as compared with the pressure pulsations in the inlet section can be obtained only in the case of using conical diffusers with the aperture angle α = 7° in which separation-free current is retained even with large inlet disturbances of the flow.

Both of the results mentioned above have a direct relation to steam turbine control valves because on one hand, they elucidate the physical nature of large pulsations of pressure downstream of the valves, and on the other hand, they clearly point to the need of using diffuser seats with aperture angles α not exceeding 8°.

RESULTS FROM THE STUDY OF THE VIBRATION STATE OF CONTROL VALVES IN FLOW WITH A HIGH INITIAL LEVEL OF PRESSURE PULSATIONS

The performed studies of the flow turned through a sharp angle have shown that, if we wish to achieve vibration reliability of control valves, we must first of all take into account that a high level of initial pressure pulsations exists in the controlled medium. In real angle valves, this level is determined not only by the disturbances resulting from turning the flow, but also by upsetting the steady nature of the flow due to an abrupt widening of the flow pass area when the working fluid enters into the valve box. The results from direct measurements of average pressure pulsations in this box obtained in the valve model (see Fig. 2) are shown in Fig. 4 as the relative average amplitude of pressure pulsations $\Delta\bar{p}$ vs. the dimensionless valve ascending height $\bar{h}$ in the cases of using diffuser seats with the aperture angles α equal to 10 and 7°.

With α increased from 7 to 10°, the value of $\Delta\bar{p}$ in the valve box shows a large growth, which points to essential influence of the flow pattern in diffuser seats on the level of flow unsteadiness in the valve box. In parallel with this, we can point out that pressure pulsations in the valve box increase with increasing the valve opening degree. It should also be noted that the frequency of pressure pulsations vary quite noticeably as $\bar{h}$ increases (Fig. 5). Thus, with $\bar{h} \approx 0.06$, high-frequency pulsations typical for highly turbulized flow were observed in the valve box (see Fig. 5a), whereas at $\bar{h} = 0.12$, low-frequency pulsations of pressure emerged along with turbulent pulsations (see Fig. 5b), which were caused by the occurrence of discrete vortex formations in the flow.

When a control valve with a subsequent diffuser seat is inserted in such flow, pressure pulsations may either increase or decrease along the flow of working medium, depending on the shapes of valve channel and diffuser seat. When the profiled channel was fitted with a diffuser seat having the aperture angle α = 10°, the relative pulsations of pressure downstream of the valve $\Delta\bar{p}$ remained almost unchanged as compared

with the pressure pulsations in the valve box (curve *3* in Fig. 4).

The situation becomes different if a diffuser seat with the aperture angle $\alpha = 7°$ is installed. In this case, the level of pressure pulsations downstream of the valve dropped by almost a factor of 2 as compared with pulsations in the valve box and increased from 2.8 to 6% as the valve was opened (curve *4* in Fig. 4). These results are in full agreement with the data from tests of angle turns with subsequent diffusers, in which the minimal pulsations of pressure in the flow downstream of the diffuser were obtained exactly at the conical diffuser aperture angles $\alpha = 7°$.

The obtained results point to the following. When aerodynamically perfect control valves are installed in unsteady flow, they do not generate additional pulsations, but on the contrary, they stabilize currents in smooth valve channels, due to which the initial level of pressure pulsations becomes noticeably smaller.

The situation becomes different in the case of using ball and disk valves, as well as balanced profiled valves with additional steam loading at high values of $\bar{h}$ when an additional steam flowrate going out from the central balancing seat is passed through the valve's inner part [5]. In this case, the initial level of pressure pulsations on the valve box does not decrease but shows a noticeable growth along the flow to reach 9–12% downstream of the diffuser seat. On the whole, this level of pulsations corresponds to the values recorded in [1, 2]. From the practical point of view, high-amplitude pulsations of pressure in the flow lead to increased vibration of the valve box and pipeline parts connected to it; in addition, they adversely affect the operation of nozzle vanes connected with the corresponding valves. In addition, if there are no efficient mechanical or aerodynamic dampers, the dynamic forces acting on the stems of control valves may become much greater owing to these pulsations and finally lead to their destruction.

As regards the vibration reliability of control valves installed in unsteady flow with a high level of pressure pulsations, a valve with high damping capacity is the best one. In this case, the level of dynamic forces acting on the stem is kept to a minimum in all positions of the valve and with all pressure differences that occur during operation.

For carrying out a comparative assessment of the dynamic forces acting on the stems, four designs of valves used in Russian turbines (Fig. 6) were studied in detail. Figure 6a shows a typical balanced valve with a ball-shaped cup, which been in operation for a long time in the turbines produced by the Leningrad Metal Works (LMZ) and in many turbines produced by the Ural Turbine Engine Works (UTMZ), and Fig. 6b shows the plate-shaped valve with additional steam loading that was designed later at the LMZ. Figure 6c shows the valve with a profiled cup, a central balancing

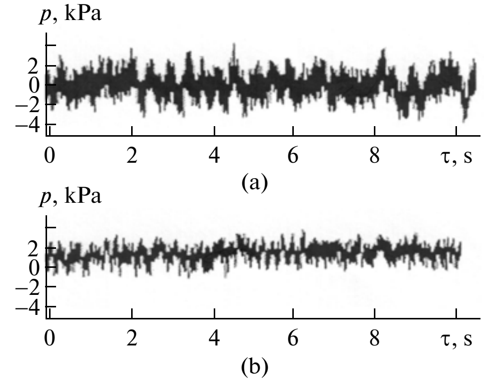

Fig. 5. Oscillograms of pressure pulsation frequencies in the valve box at $\bar{h} = 0.06$ (a) and $\bar{h} = 0.12$ (b).

hole, and zero-flow additional steam loading at high ascending heights (Fig. 6d) that was developed by the ENTEK Co. jointly with the Moscow Power Engineering Institute.

All these valves were tested on the installation (see Fig. 1) with the pressure ratios $\varepsilon_2 = 0.50–0.96$ ($\varepsilon_2 = p_2/p_0$, where p_0 and p_2 are the initial pressure in the valve box and air pressure downstream of the diffuser seat) in the self-similarity region with respect to Reynolds number. Valves with the same diffuser seats and aperture angle $\alpha = 7°$ were studied. In addition, we studied the valve designed jointly by specialists of ENTEK and Moscow Power Engineering Institute that was fitted with a diffuser seat and had the aperture angle $\alpha = 10°$. The most characteristic oscillograms of forces on the stems obtained in these tests are shown in Fig. 7.

As was expected, the maximal amplitude of force pulsations was recorded on the stem of the ball valve (see Fig. 7a). At high valve opening degrees ($\bar{h} = 0.25$ and $\varepsilon_2 = 0.926$), the pulsations of forces were commensurable with the static loads acting on the stem. The obtained result is explained by the fact that pressure pulsations in the valve box superimposed on the pressure pulsations caused by unsteady separation of flow from the valve cup's ball surface. Lack of axial symmetry in the case of working medium being laterally admitted to angle valves must be taken into account here. As a result, the line along which the flow separates from the ball surface is also unsymmetrical and not fixed in space due to high unsteadiness of the flow.

The plate-type valve (Fig. 6b), the flow in which separates over the entire end-face perimeter of the valve, and a jet current with high-frequency pulsations of all parameters of the flow settles downstream of it, is free from the above-mentioned drawback. However, at high opening degrees, the valve is loaded by an additional force resulting from increased flowrate of steam through the internal part. As a result, an essentially

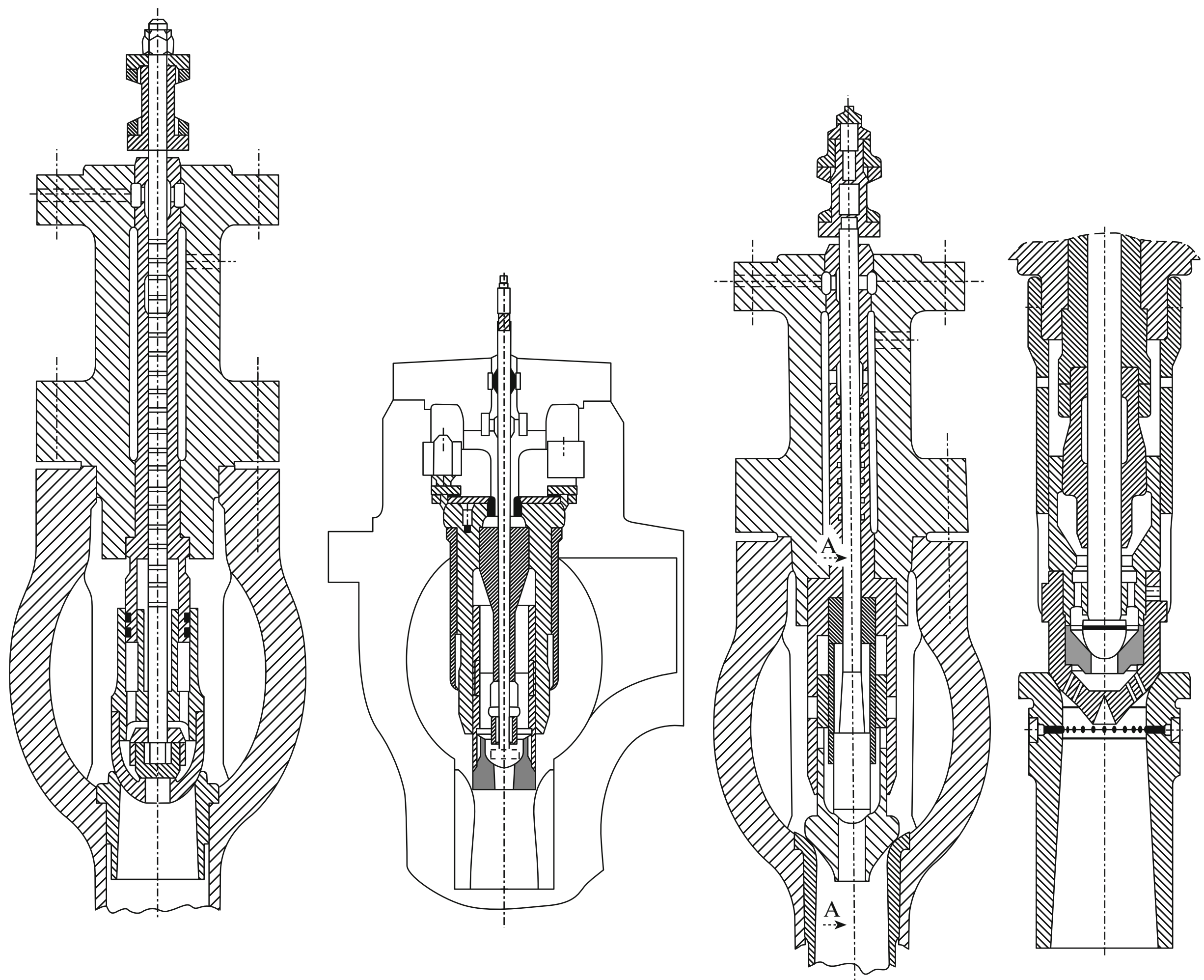

Fig. 6. Designs of valves installed in Russian turbines.

higher steam flowrate through the balancing seat's central hole is passed. Additional low-frequency pulsations of pressure appear downstream of the when the main flow is mixed with the secondary one. According to [1], the absolute pulsation of pressure reaches 10–15% of the initial steam pressure. If there are no aerodynamic dampers, this leads to the occurrence of high dynamic forces on the stem (see Fig. 7b). The dynamic forces applied to the stem of the profiled valve designed by Energotekh with central balancing and flow-type additional steam loading (see Fig. 6c) were found to be smaller at high values of $\bar{h}$ (Fig. 7c), but these forces act at almost all positions of the valve.

A fundamentally different picture was recorded in the balanced profiled valve designed jointly by specialists of ENTEK and the Moscow Power Engineering Institute (Fig. 6d), which has a perforated streamlined surface, damping chamber, and zero-flow steam loading [4]. Almost no dynamic forces are applied to the stem of this valve (see Fig. 7d) at all positions of the valve and at all studied pressure ratios ε_2. The main feature in which the considered balanced valve differs from all other ones is that working medium is removed from the valve balancing system through the perforation holes near the setdown diameter and not through the central hole in the valve cup. In addition, a special chamber is placed between the balancing valve's outlet seat and the space downstream of the valve, which plays the role of an aerodynamic damper, and the valve is additionally loaded at large ascending heights as a result of closing steam admission to the balancing seat. Therefore, the inner leak of steam at $\bar{h} > 0.15$ is reduced to its minimal value and does not upset the stability of flow in the valve channel.

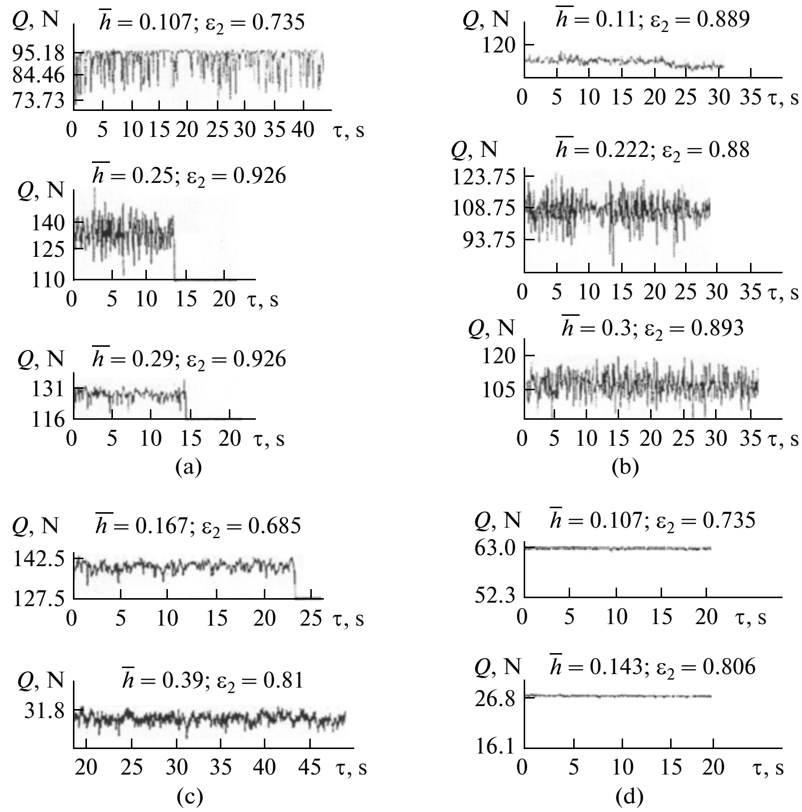

Fig. 7. Typical oscillograms of forces Q applied to the stems of studied valves.

Studying the valve the diffusion seat of which has an aperture angle equal to 10° is of much interest for practical applications. As was already noted, increased pulsations of pressure occur in the flow when such seat is used in control valves, and the occurrence of additional dynamic forces on the stem can be expected.

However, insignificant pulsations of forces on the stem were revealed during the tests only at $\bar{h} \approx 0.10$–0.154. As earlier, no dynamic forces were revealed at other values of $\bar{h}$, which points to high efficiency of the used damper. At the same time, a very large difference in the values of parameters was revealed from the measurements of vibration displacements on the valve box shell. The vibration displacements of the shell measured on the valve with the seat's diffuser aperture angle $\alpha = 7°$ varied only slightly with increasing $\bar{h}$ and were equal to around 10 μm in the model, whereas at $\alpha = 10°$ their value grew from 10 to 35–38 μm with increasing $\bar{h}$. In other words, a growth of pressure pulsations in working medium inevitably entails higher vibration of the entire control valve; however, this growth has little effect on the dynamic forces on valve stems if the valves are fitted with efficient aerodynamic dampers.

CONCLUSIONS

(1) High pulsations of pressure in the working medium downstream of steam turbine control valves are determined not only by their design features, but to a considerable degree by the flow becoming unstable when it is turned through 90° with respect to its initial direction.

(2) The vibration state of control valves is stemming from the degree of their interaction with unsteady flow characterized by a high level of pressure pulsations. If the valve is fitted with efficient aerodynamic dampers, this interaction can be kept to a minimum and have not any noticeable effect on the vibration reliability of the valve itself.

(3) If a valve is fitted with efficient aerodynamic dampers, the level of pressure pulsations in working medium should not be used as an indicator characterizing the reliability of such valve.

(4) The vibration state of valves depends essentially on the aperture angle of the diffuser seats used in them. Seats the diffuser part of which has the aperture angle $\alpha = 7°$ are the optimal ones in this respect. Increasing α by as small as 3° (to 10°) leads to a very high growth of vibration displacements of all external parts of the valve.

REFERENCES

1. A. G. Kostyuk, A. I. Kumenko, A. S. Nekrasov, and S. V. Medvedev, "An Experimental Analysis of Pressure Pulsations in the Steam Admission Elements of a Turbine Installation," Teploenergetika, No. 6, 50–57, 2000 [Therm. Eng., No. 6, 529 (2000)].
2. V. F. Kasilov, S. V. Kalinin, V. M. Gvozdev, et al., "A Study of the Vibrational Activity of the Control Valves in the Steam Admission System of the HP Cylinder in a K-200-130 Steam Turbine," Teploenergetika, No. 11, 13–20, 2001 [Therm. Eng., No. 11, 890 (2001)].
3. L. Tajč and L. Bednář, Št'astny Regulačni Ventily pro Turbiny Veukého Vykony Energeticke Stroje (Zapadoceska Univerzita v Pezni, Června, 2003), pp. 101–107.
4. A. E. Zaryankin and B. P. Simonov, *Control and Stop-and-Control Valves of Steam Turbines* (MEI, Moscow, 2005) [in Russian].
5. A. M. Gotovtsev, A. E. Zaryankin, S. V. Arianov, and A. N. Paramonov, "Comparative Assessment of the Vibration State of the Control Valves Installed in K-200-12.8 Turbines of LMZ," Teploenergetika, No. 12, 59–66, 2005 [Therm. Eng., No. 12, 968 (2005)].